Investigation of Effect of Stratification on the Thermal Performance of Packed Bed Solar Air Heater

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Abstract

Use of rock-beds with bed elements having geometrical shapes and sizes to result in temperature stratification for the improvement of performance of solar air heater has been proposed by several investigators. However, this enhanced efficiency is accompanied by substantial increase in pressure loss, so the solar energy storage system should be optimized in such a way that there is maximum thermal gain accompanied by minimum possible pressure losses. A computational analysis of temperature distribution of the bed elements of the storage system as a function of time and location has been determined. The resulting stratification as a function of sphericity (Ψ), void fraction (ε) and equivalent diameter (D_e) of the bed elements and operating parameters namely temperature rise parameter ($\Delta T/I$) and insolation (I) has been determined. Optimum values of bed parameters namely sphericity (Ψ), void fraction (ε) and equivalent diameter (D_e) as a function of temperature rise parameter and insolation have been estimated on the basis of Maximum effective efficiency (η_e) of the solar air heater. These values can be used by a designer to select the optimum values of bed parameters.

Keywords

Equivalent Diameter; Sphericity; Stratification; Temperature Rise Parameter; Void Fraction

Introduction

Solar air heating systems which utilize rock-beds for energy storage, are quite common. Harmeet Singh (2010) reviewed the packed bed solar energy storage systems. The overall performance of such systems is influenced significantly by the temperature distribution in the rock-bed energy storage unit which is strongly affected by the system parameters i.e. sphericity, void fraction and equivalent diameter of the bed. Duffie and Beckman (1991) described that the well designed packed bed having several characteristics which are desirable for solar energy application such as high heat transfer between the air and solid promotes thermal stratification. Heat transfer behavior of the packed bed was studied by Schumann (1929) in the form of mathematical model. A. Mawire and Mc Pherson (2009) simulated the performance of storage materials for pebble bed thermal energy storage (TES) systems. Common beds employed for energy storage have been found to have high pressure drop in the bed, resulting in substantially large energy consumption to propel air through the bed. This reduces the overall benefit of the solar energy utilization system. Pressure drop in the bed could be reduced with the use of large sized bed element. Reduction in the heat transfer rate to large size material element due to smaller surface area per unit volume of storage is compensated by substantial reduction in the amount of energy consumption by fan due to low pressure drop in the bed. It can therefore be beneficial to use large size materials as the storage element. Nusselt number and friction factor correlations for packed bed solar energy storage system having large sized elements of different shapes have been developed by Singh (2005).

A simple fluid and heat flow analysis for a packed bed has been presented by Howell et. al.(1982). The rate of heat transfer to and from the solid in the packed bed is a strong function of physical properties of the fluid and solid. The local temperature of the fluid and the surface of the solid, flow rate of the fluid and the physical characteristics of the bed have been stated by Demirel and Kaharaman (2000). Loff and Hawley (1948) have given simple correlation for volumetric heat transfer coefficient between air and bed element. Mumma and Marvin (1976) proposed a straight forward simulation method for the prediction of behavior of the packed bed element. A theoretical model by W. F. Phillips (1981) can predict the effect of stratification in the rock bed storage unit of a solar heating system.

The benefit of additional thermal energy gain as well as the additional energy input to overcome pressure loss has been simultaneously considered. Cortes and Piacentini (1990) have proposed a parameter, known as effective efficiency defined below:

$$\eta_e = \frac{Q_u - \frac{\Delta p\ a}{C}}{I A_c} \times 100 \quad (1)$$

where, Q_u is useful energy gain in the collector (W), ΔP is pressure drop across the bed, $\square_a$ is mass flow rate, I is the insolation (W/m^2), A_c is the collector area (m^2) and C is the conversion factor that represents the ratio of mechanical energy required for pump work to the amount of thermal energy that will be required to produce that work (C=0.2), such that the quantity ΔP m_a/C is the equivalent thermal energy to produce the required pumping energy.

In this work, the performance of a solar collector sensible heat storage system composed of large sized element has been predicted. Thermal and effective efficiency has been determined as function of system and operating parameters. The effect of parameters on the stratification and the thermal performance of solar collector have been discussed. The optimum values of system parameters have been calculated that yield the maximum effective efficiency for given set of value of operating parameters.

Solar Energy Collector-Storage System

Fig. 1 shows the schematic of the solar energy cum storage system that has been analyses. The solar collector is the typical flat plate solar air heater with single transparent cover and a non-selective coating suitable for low temperature air heating whereas the storage consists of large size packing elements typically analysed by Singh Ranjit et. al. (2006) for which correlation for heat transfer coefficient and friction factor are available. Collector is assumed to have a control system such that it supplies a constant temperature outlet temperature for a variable inlet temperature which is the outlet temperature of the storage tank, thus resulting in variable air flow rates through the collector as well as the storage system.

Fig. 2 shows the geometrical characteristics of the storage elements. Further, the solar energy utilization system design is based on the following important operating of design parameters.

Temperature rise (ΔT) represents the temperature difference between outlet temperature of the collector and the ambient temperature. This rise typically represents the type of applications; and each application requires specific outlet temperature from the collector i.e a temperature rise brought about by the collector if the inlet temperature was ambient air temperature.

Average insolation (I) represents the incoming solar radiation averaged of the period of utilization and is evaluated for the place where the system is located.

Typically, the ratio ΔT/I, known as temperature rise parameter and insolation(I) are given the name of operating or design parameter of the solar energy utilization system.

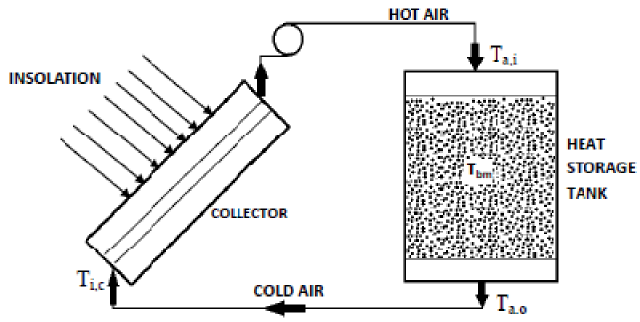

FIG. 1 SCHEMATIC OF THE STORAGE SYSTEM UNDER CONSIDERATION

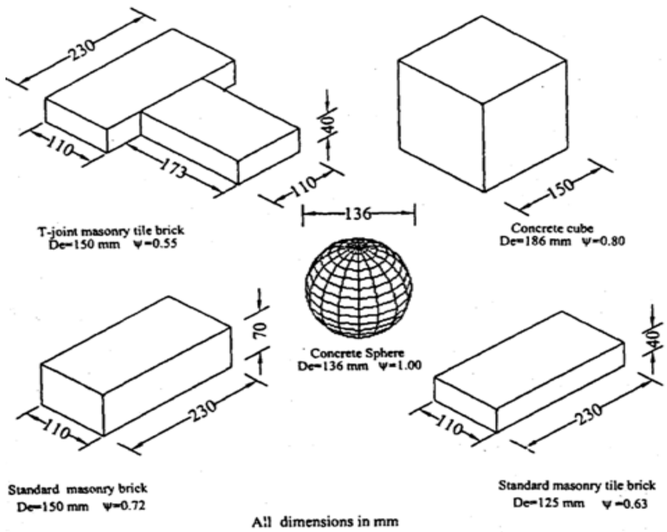

FIG. 2 DIMENSIONS, EQUIVALENT DIAMETER (DE) AND SPHERICITY(Ψ) OF DIFFERENT MATERIALS

Fixed and Variable Parameters

Configurationally, parameters of the solar collector, storage element parameters and operating parameters can be divided into fixed and variable parameters of the system that has been considered for analysis. These are listed in Tables 1, 2 and 3.

TABLE 1 FIXED PARAMETERS OF BED

Description	Parameter	Value
Volume of packed bed, m^3	(V_b)	15
Length of packed bed, m	(L)	6
Number of bed element	(N)	60
Initial bed temperature, °C	(T_{bi})	25
Dynamic viscosity of air, kg/s-m	(μ_a)	1.865×10^{-5}
Density of air, kg/m^3	(ϱ_a)	1.1
Inlet air temperature to bed, °C	T_{ai} or T_{bi}	40
Ambient temperature, °C	T_∞	25
Density of storage material, kg/m^3	ϱ_s	1920
Specific heat of air, (J/kg°C)	C_{pa}	1008
Specific heat of storage material, J/kg°C	C_{ps}	835
Collector area, m^2	A_c	20
(for collector)	$F_R(\tau\alpha)_e$	0.62
(for collector)	F_RU_l	3.38
Time interval, min.	Δt	1

TABLE 2 VARIABLE PARAMETERS OF BED

Equivalent diameter of packing material,(m)	D_e (m)	Corresponding to the material element under consideration. 0.05-0.2
Sphericity of material element	Ψ	Corresponding to the material element under consideration. 0.5-1
Void fraction	ε	Corresponding to the material element under consideration. 0.30-0.50
Inlet to bed temperature	Ti (°C)	35°C to 75 °C
Insolation	I(W/m^2)	500, 750, 1000
Temperature difference	ΔT	10°C to 55°C
Temperature rise parameter	$\Delta T/I$ (°Cm^2 /W)	0.05 to 0.1

TABLE 3 RANGE OF SYSTEM PARAMETERS

Parameter	Range
Sphericity, ψ	0.5 to 1
Void fraction, ε	0.3 to 0.5
Equivalent diameter, D_e (cm)	5 to 20

Prediction of Performance

The performance of the system with respect to thermal and effective efficiency of the collector has been predicted on the basis of detailed consideration of heat and fluid flow processes in the storage-collector system shown in Fig. 1. The calculation starts with a set of values of operating parameters ($\Delta T/I$, and I) and proceeds with the determination of efficiencies for all possible sets of geometrical parameters of the storage bed element(ε,Ψ and D_e). In this process, the following sets of well known relationship are utilized:-

Useful thermal energy gain, Q_u, by using Hottle and Willier (1958) equation.

$$Q_u = A_c \times [\,(F_R(\tau\alpha)_e \times I) - F_R U_l (T_{ic} - T_{amb})] \tag{2}$$

where F_R is the heat removal factor defined as ratio of actual useful energy gain to the maximum possible energy gain, $(\tau\alpha)_e$ represents effective transmittance-absorptance product, U_l is overall heat loss coefficient (W/m^2 ^{0}C) and T_{ic} is the outlet temperature of the air from the packed bed.

Temperature distribution of bed and air, by means of Mumma and Marvin (1976) for the storage bed shown in Fig. 3.

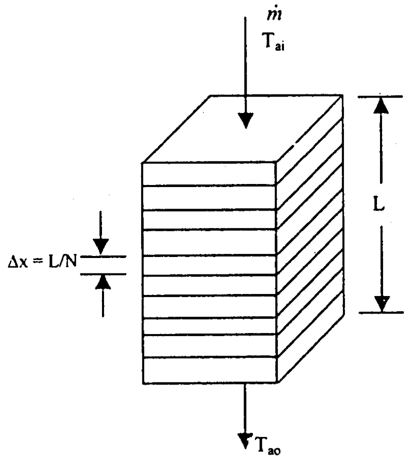

FIG. 3 (a) PACKED BED

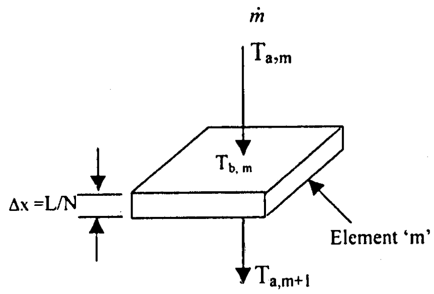

FIG. 3 (b) ELEMENT 'M' OF BED

The final temperature of the air '$T_{a,m+1}$' at the exit of each element is:

$$T_{a,m+1} = T_{b,m} + \left(T_{a,m} - T_{b,m}\right) \exp(-\Phi_1) \tag{3}$$

where $\Phi_1 = \frac{NTU}{N}$ and NTU (number of transfer unit)= $\frac{h_v A_b L}{(C_p)_a}$ where A_b is cross sectional area of packed bed, L is length of packed bed, N is number of bed elements.

Volumetric heat transfer coefficient, 'h_v' is calculated by using the expression given by Chandra and Willits (1981) for Nusselt number in the modified form as:

$$Nu = \frac{h_v D_e^2}{k_a} \quad (4)$$

where k_a is thermal conductivity of air. The correlation for Nusselt number is given by Singh, Ranjit (2006)

$$Nu=0.437\times R_e^{0.75}\times\psi^{3.35}\times\varepsilon^{-1.62} \times\exp(29.03\times\log10(\psi)^2))/D_e^2] \quad (5)$$

Thus the expression for volumetric heat transfer coefficient is:

$$h_v=k_a \times 0.437 \times Re^{0.75}\times \Psi^{3.35}\times\varepsilon^{-.62} \times[\exp(29.03(\log \Psi^2))]/D_e^2 \quad (6)$$

The temperature of the bed element '$T_{b,m(t+\Delta t)}$' is given by:

$$T_{b,m(t+\Delta t)} = T_{b,m(t)} + \left[\Phi_2\left(T_{a,m} - T_{a,m+1}\right) - \Phi_3\left(T_{b,m} - T_{amb}\right)\right] \quad (7)$$

where $\Phi_2 = \frac{(C_p)_a N}{(\varrho C_p)_s AL(1-\varepsilon)}$ and $\Phi_3 = \frac{(U\Delta A)_m}{(C_p)_a}\Phi_2$ and Δt is time increment.

Mass flow rate of air from:

$$m_a=Q_u/C_{pa}(T_{ib}-T_{ic}) \quad (8)$$

where T_{ib} is inlet to bed temperature and T_{ic} is the inlet to collector temperature.

Pressure drop in the bed (ΔP) has been calculated by using the following friction factor correlation given by Singh, Ranjit. (2006)

$$f = 4.466(R_e)^{-0.2}(\psi)^{0.696}(\varepsilon)^{-2.945}[\exp\{11.85(\log\psi)^2\}] \quad (9)$$

thus $\Delta P= LG^2 4.466 R_e^{-0.2}\psi^{0.696}\varepsilon^{-.945} \times\exp(11.85(\log_{10}\psi^2))/(\varrho_a D_e) \quad (10)$

Thermal efficiency of collector is determined as the ratio of useful energy gain to the incident solar energy.

$$\eta_{th} = \frac{Q_u}{IA_c} \quad (11)$$

where Q_u is useful heat gain by collector. The average thermal efficiency of the collector is calculated using the following relation:

$$\eta_{th,avg} = \frac{\Sigma_0^n Q_u}{IA_c\Delta t*3600*8}\times 100 \quad (12)$$

Where 'n' is a counter defining number of observations during the complete charging time.

The effective efficiency, η_e is calculated as follows;

$$\eta_e = \frac{Q_u - \frac{\Delta p\, a}{C}}{IA_c} \times 100 \quad (13)$$

The expression for the average effective efficiency is as follows:

$$\eta_{e,avg} = \frac{\Sigma\left(Q_u - \frac{\Delta p\, a}{C}\right)}{IA_c*3600*8}\times 100 \quad (14)$$

where ΔP is pressure drop across the bed, □a is mass flow rate and C is the conversion factor to account for conversion of high grade mechanical energy to thermal energy (C=0.2).

Phillips W. F (1981) gave following expression for Stratification coefficient, K_s :

$$K_s = \frac{\ln[1/(1-E)]}{\{1+M\ln[1/(1-E)]\}E} \quad (15)$$

Thus it is seen that K_s depends on three dimensionless numbers, E, M and NTU.

here, Heat exchanger effectiveness (for collector),

$$E = \frac{F_R A_c U_l}{C_{pa}} \quad (16)$$

Mixing number., $M = M_k + \frac{1}{NTU}$ (17)

Conductivity number, $M_k = \frac{A_c k}{C_{pa} L}$ (18)

No. of transfer unit, $NTU = \frac{h_v V_b}{\dot{m} C_{pa}}$ (19)

Result and Discussion:

The results of mathematical simulation of the system have been discussed in the system. The effect of system parameters (Sphericity Ψ, void fraction ε and equivalent diameter D_e) on the performance parameters, namely thermal and effective efficiency has been investigated. In view of the fact that the temperature distribution in the bed is the most important aspect, it is beneficial to discuss the effect of system and operating variable on temperature distribution.

Temperature Distribution and Thermal Performance of Collector

Mathematical simulation has been used to yield the average temperature of each element of storage system considered, during charging at a given instant of time. Fig. 4 shows the effect of void fraction (ε) on the temperature distribution. It can be observed that the stratification reduces with the increase of void fraction. The temperature of bottom most bed element being higher leads to higher outlet air temperature. The mean bed temperature was observed to increase with an increasing void fraction. These variations are reflected in the outlet temperature of air which is inlet temperature of solar collector as can be seen in Fig. 5 where the collector inlet temperature has been plotted as a function of charging time for different values of void fraction. The effect of these temperature variations on the collector efficiency can be seen in Fig. 6 where thermal efficiency of collector has been plotted

as a function of void fraction for different values of charging time. It is seen that the lowest void fraction results in the highest value of average thermal efficiency.

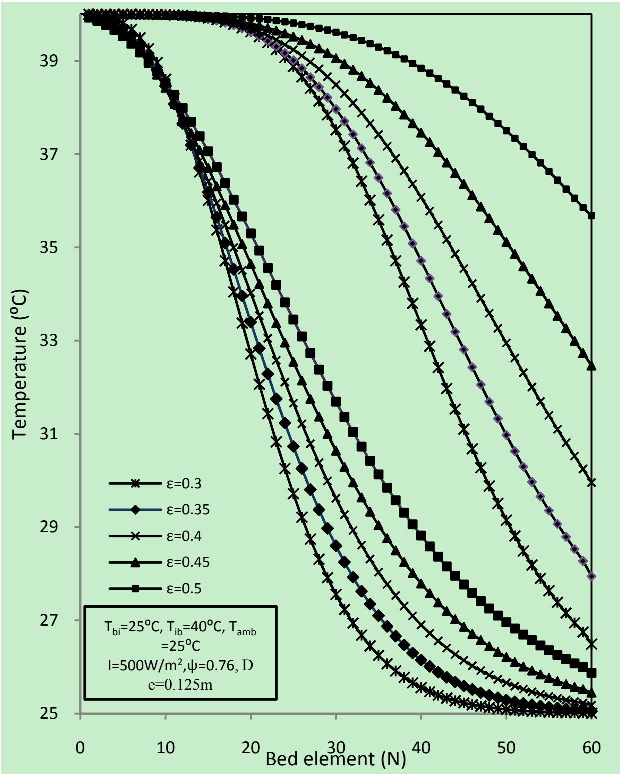

FIG. 4 EFFECT OF CHARGING OF THE BED ON INLET AIR TEMPERATURE TO THE COLLECTOR AT DIFFERENT VOID FRACTION OF THE BED

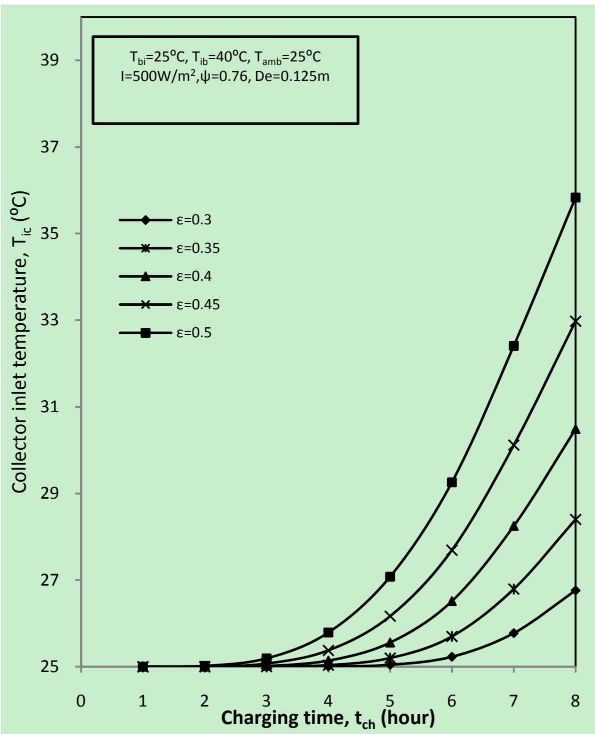

FIG. 5 EFFECT OF CHARGING OF THE BED ON INLET AIR TEMPERATURE TO THE COLLECTOR AT DIFFERENT VOID FRACTION OF THE BED

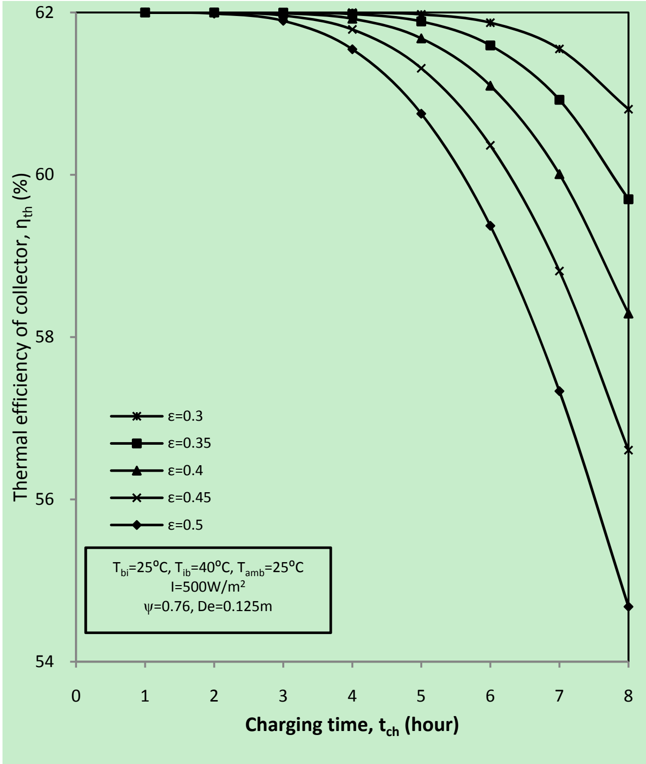

FIG. 6 VARIATION OF THERMAL EFFICIENCY OF COLLECTOR WITH CHARGING OF BED AT DIFFERENT VOID FRACTION

Similarly, calculations for the effect of sphericity and equivalent diameter of the bed element were carried out and it was found that in general, a sphericity of 1.0 and the lowest value of equivalent diameter resulted in the best thermal performance of the collector.

Thermohydraulic Performance

As pointed out earlier, any effort at improving the heat transfer performance of the system is usually accompanied by additional frictional losses; which becomes essential to select the system parameters such that the system yields maximum possible enhancement in thermal performance with minimum possible enhancement in friction losses. This can be done by considering these two effects simultaneously through thermohydraulic performance i.e. the effective efficiency.

Fig. 7 shows the plot of average effective efficiency as a function of void fraction and temperature rise parameter. An optimum value of void fraction is one that results in maximum values of effective efficiency for a fixed value of temperature rise parameters ($\Delta T/I$, and I). as it can be seen that for a low value of temperature rise parameter, the lower value of void fraction results in maximum value of effective efficiency whereas the higher value of void fraction yields maximum value of efficiency at higher value of temperature rise parameter. It has also been found that

the optimum values of void fraction are also function of insolation (I). Similar procedure was employed to evaluate the optimum values sphericity and equivalent diameters as a function of temperature rise parameter and insolation.

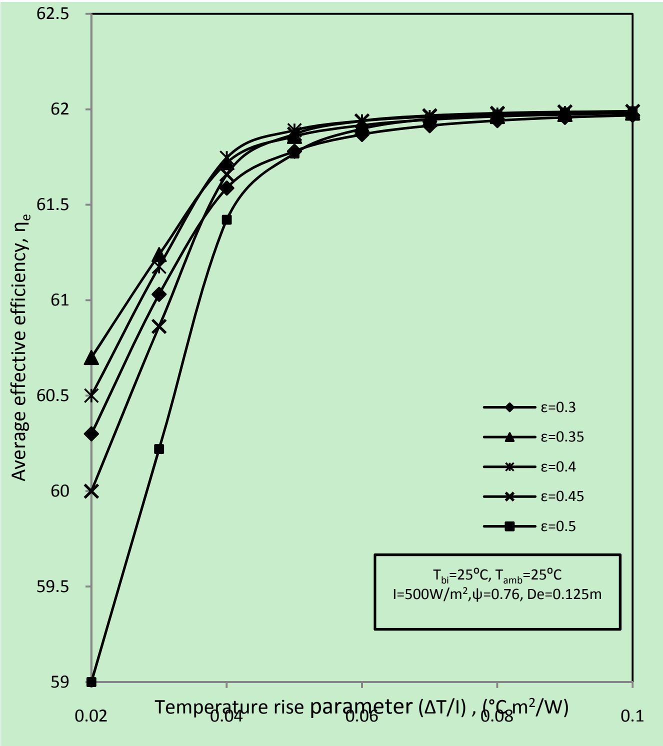

FIG. 7 EFFECT OF TEMPERATURE RISE PARAMETER ON AVERAGE EFFECTIVE EFFICIENCY FOR DIFFERENT VALUES OF VOID FRACTION

Stratification Coefficient

It is well known that the performance of a collector system receiving air coming out from the bed is considerably influenced by the stratification of the bed. As the charging time increases, the stratification coefficient decreases as the temperature of the bed increases. Stratification coefficient is high at the beginning of charging due to higher temperature gradient, but with the passing of time, the bottom layer of the pebble bed gets heated up, leading to decrease in the temperature gradient and then lowering stratification coefficient. The variation of average stratification coefficient with void fraction is shown in Fig. 8. The increase in void fraction value decreases the stratification coefficient. The bed at void fraction of 0.3 has the maximum stratification. At void fraction of 0.50, the bed is least stratified. This is due to the fact that with increase of void fraction the heat transfer coefficient between the air and solid decreases and vice versa. Fig. 9 shows the variation of collector efficiency with stratification coefficient, and the efficiency of solar collector is high for higher value of stratification coefficient as the temperature gradient is high(temperature at top of pebble bed is high and temperature at bottom layer is low) thus enhancing the collector efficiency.

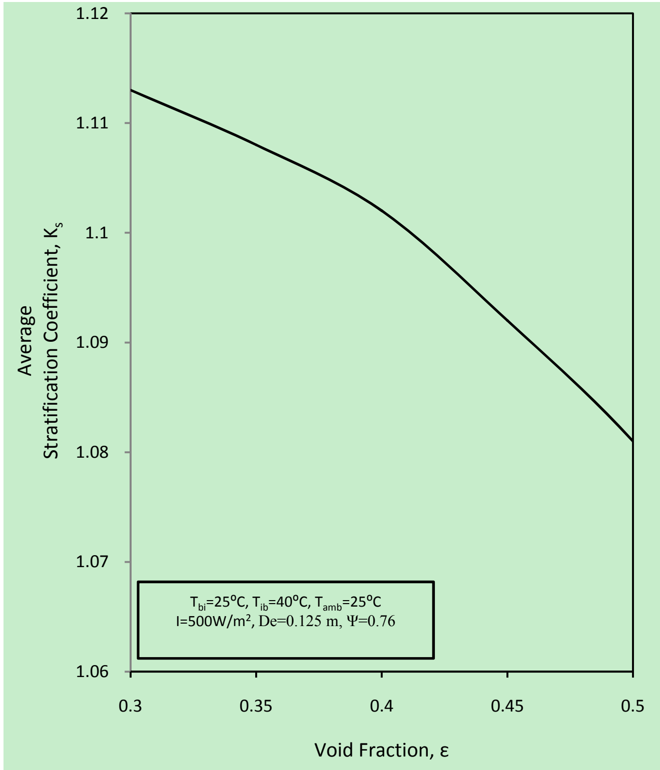

FIG. 8 VARIATION OF AVERAGE STRATIFICATION COEFFICIENT WITH VOID FRACTION

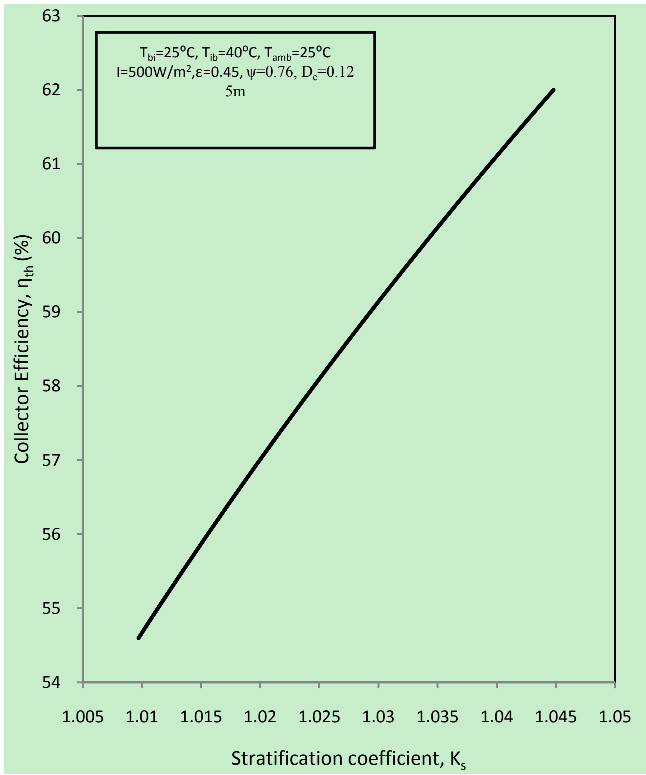

FIG. 9 VARIATION OF COLLECTOR EFFICIENCY WITH STRATIFICATION COEFFICIENT

Optimum Values of System Parameters

In order to determine the set of values of sphericity, void fraction and equivalent diameter out of all investigated sets (i.e. for different values of sphericity, void fraction and equivalent diameter) that yield maximum value of effective efficiency of collector for a given set of design conditions (i.e. temperature rise parameter, ΔT/I and insolation, I), the effective efficiencies of corresponding individual sets were compared. From this comparison, the set that yielded maximum effective efficiency value for given set of design condition was determined. Tables 4, 5 and 6 list the optimum sets; the corresponding system parameters (sphericity, void fraction and equivalent diameter) for different values of temperature rise parameters (from 0.01 to 0.1) and insolation (I = 500 W/m^2 to 1000 W/m^2). These optimum values have also been represented in Figs. 10, 11 and 12.

TABLE 4 VALUES OF OPTIMAL SYSTEM PARAMETERS FOR I = 500 W/m^2

ΔT/I (Temperature rise parameter)	Optimum Sphericity (ψ)	Optimum Void fraction (ε)	Optimum Equivalent diameter (D_e), m
0.01	1	0.35	0.05
0.02	1	0.35	0.05
0.03	1	0.35	0.05
0.04	1	0.4	0.05
0.05	1	0.4	0.05
0.06	1	0.4	0.05
0.07	1	0.45	0.05
0.08	1	0.45	0.05
0.09	1	0.45	0.05
0.1	1	0.45	0.05

TABLE 5 VALUES OF OPTIMAL SYSTEM PARAMETERS FOR I = 750 W/m^2

ΔT/I (Temperature rise parameter)	Optimum Sphericity (ψ)	Optimum Void fraction (ε)	Optimum Equivalent diameter (D_e), m
0.0133	1	0.35	0.05
0.02	1	0.35	0.05
0.0265	1	0.4	0.05
0.0333	1	0.4	0.05
0.04	1	0.4	0.05
0.0466	1	0.4	0.05
0.0533	1	0.45	0.05
0.06	1	0.45	0.05
0.066	1	0.45	0.05
0.0733	1	0.5	0.05

TABLE 6 VALUES OF OPTIMAL SYSTEM PARAMETERS FOR I = 1000 W/m^2

ΔT/I (Temperature rise parameter)	Optimum Sphericity (ψ)	Optimum Void fraction (ε)	Optimum Equivalent diameter (D_e), m
0.015	1	0.4	0.05
0.02	1	0.4	0.05
0.025	1	0.4	0.05
0.03	1	0.45	0.05
0.035	1	0.45	0.05
0.04	1	0.45	0.05
0.045	1	0.45	0.05
0.05	1	0.5	0.05
0.055	1	0.5	0.05
0.06	1	0.5	0.05

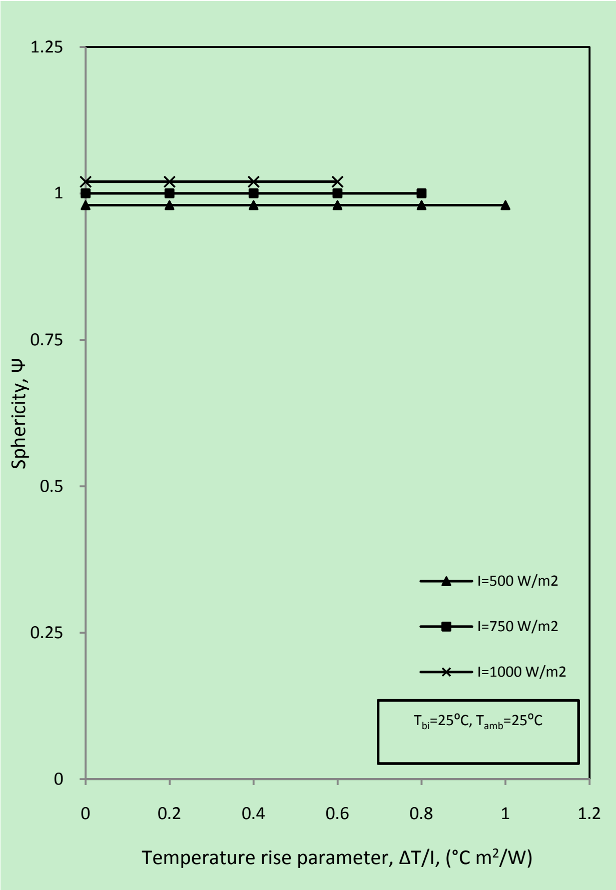

FIG. 10 OPTIMUM VALUES OF SPHERICITY FOR DIFFERENT VALUES OF INSOLATION

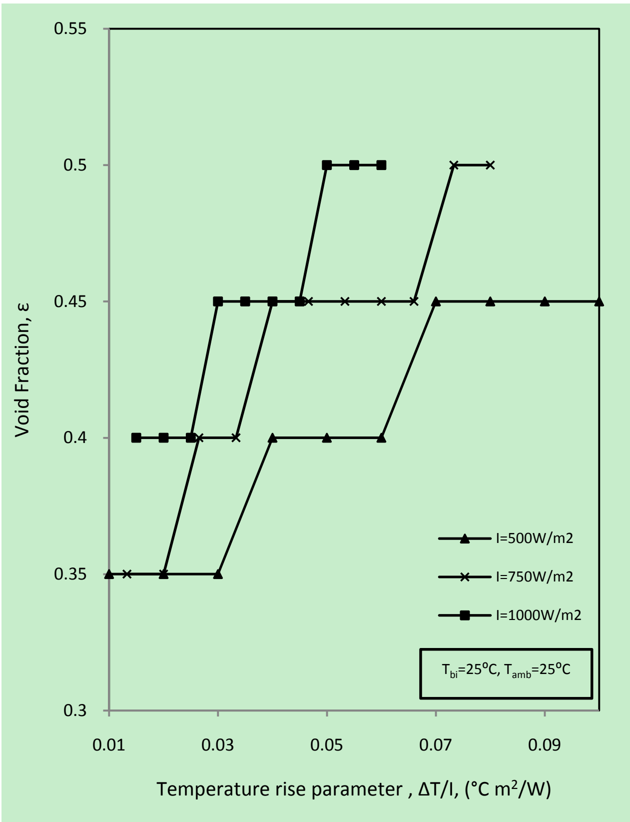

FIG. 11 OPTIMUM VALUES OF VOID FRACTION FOR DIFFERENT VALUES OF INSOLATION

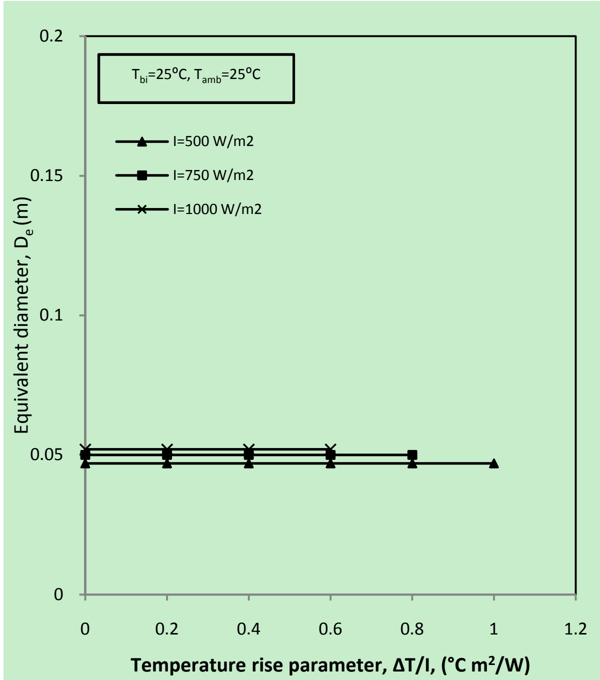

FIG. 12 OPTIMUM VALUES OF EQUIVALENT DIAMETER FOR DIFFERENT VALUES OF INSOLATION

The designer can select the optimum values of sphericity, equivalent diameter and void fraction as a set for the packed bed solar energy storage system from the plots shown in Figs. 10, 11 and 12 respectively as a function of temperature rise parameter $\Delta T/I$ and insolation I. For instance, for a location with an average insolation of 500 W/m^2 (say) and for an application requiring a temperature rise of 15°C (say) above ambient; the design parameters can be estimated as:

$\Delta T/I$= 15/ 500 =0 .03 °C m^2/W, I= 500, W/m^2

Using tables or figs, as mentioned above, the optimum bed parameters are determined as: Sphericity: 1, Void fraction: 0.35, Equivalent diameter: 0.05.

Conclusion

A solar energy storage-cum-collection system using large size bed element has been analyzed and it has been found that the effective efficiency is a strong function of geometrical parameters of the storage bed and optimum set of these parameters is a function of operating conditions. The optimum sets have been determined for different values of design condition, which can be used by the designer to obtain minimum possible friction losses in the storage system.

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